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OPERATIONAL MODAL ANALYSIS APPLICATION FOR THE MEASURE OF LOGARITHMIC DECREMENT IN CENTRIFUGAL COMPRESSOR

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ABSTRACT

The results of a stability test on a centrifugal compressor during the full load test of the entire compression train are presented.

A Magnetic Exciter was installed at the shaft end in order to excite the first lateral mode of the rotor at running conditions, the frequency and logarithmic decrement of the first mode was identified by analyzing the Frequency Response Functions with classic techniques (D.J. Ewins, 1984).

A new method for turbomachinery applications based on Operational Modal Analysis (OMA) is presented for the identification of modal parameters. Thanks to the natural excitation exerted by the gas flow on the rotor it is possible to avoid the use of any external device to perform the identification. The vibration signals have been recorded over proper measurement time windows with the compressor at steady state. The Stochastic Subspace Identification (a state-space algorithm in time domain) has been used to post-process the data. Modal parameters of the first lateral mode (natural frequency, damping ratio and mode shape) have been identified.

A comparison between the two identification techniques on the identified frequency and logarithmic decrement of the first lateral mode was performed showing the potential of the OMA methodology.

In addition four numerical cases with different stability levels were studied to confirm the reliability of the methodology for rotating machinery. A transient analysis was performed by applying a random excitation at mid-span and vibration responses have been evaluated at bearing locations. For each case the modal parameters were identified using OMA and compared with the exact theoretical values.

INTRODUCTION

Stability analysis of centrifugal compressors should ensure stable operations of the machine at running conditions. API Level 2 stability analysis investigates the dynamic behavior of rotor including the modelling of supports compliance, balance drum gas forces and inter-stage seals forces.

Although the modeling of destabilizing effects has significantly improved in the last years, uncertainties on bearing, labyrinth and oil seals rotordynamic parameters, as shown by Kocur et al. (2007), make the computational stability assessment a challenge. Given these uncertainties, end users and original equipment manufacturers (OEMs) may choose to experimentally measure a machine's stability. Moreover, the quantitative knowledge of the stability of the machine by the measure of the logarithmic decrement and natural frequency of the system allows the assessment on how conservative are the OEM's predictions, thus, allowing pushing for a more aggressive design.

The conventional stability test makes use of an external device assembled to a shaft end of the machine in order to apply a known force to the laminations present on the rotor. This force can be controlled in frequency, direction and magnitude. The measurement of the radial vibrations and the exciting force allow the evaluation of the Frequency Response Function (FRF). Using modal identification methodologies, Cloud et al (2009), FRF can be used to identify the frequency and logarithmic decrement of the first lateral mode.

The drawback of traditional stability test is that the shaft end of the rotor needs to be modified in order to have enough space for the magnetic exciter. An additional limitation in the use of traditional stability test is the case of passing through machines where the exciter cannot be easily installed.

Operational modal analysis is a technique that allows the execution of a stability test without requiring any external force. It has been widely used for the modal identification of large structures where the excitation is provided by the environment (e.g. wind or traffic) and where the application of an external load would be challenging.

In turbomachinery, the excitation source is generated by the interaction between the gas flow and compressor components. These forces need to have a sufficient energy level and frequency content (broad band and as close as possible to zero mean white noise) to excite the first lateral mode of the rotor in order to be identified by OMA. The harmonics linked to the rotational speed provide the dominant excitation, and therefore appropriate techniques shall be applied to separate the modes of interest.

Apart from the stability assessment, this method can be also successfully applied for troubleshooting purpose, for example, an expert user can recognize if a sub-synchronous vibration is due to a forced response phenomena or it is due poorly damped 1st mode.

This paper presents a comparison of the modal parameters identified by using OMA and by applying traditional stability test technique during the full load test of a high pressure centrifugal compressor equipped with a magnetic exciter. The details of the tested machine are reported in Table 1. The cross section of the machine is shown in Figure 1.

Type	BCL306	
Rating	/C	
Stage number	6	
Suction pressure	249.5	bar
Discharge pressure	550	bar
Maximum continuous speed	10857	RPM

Table 1: The BCL306/C

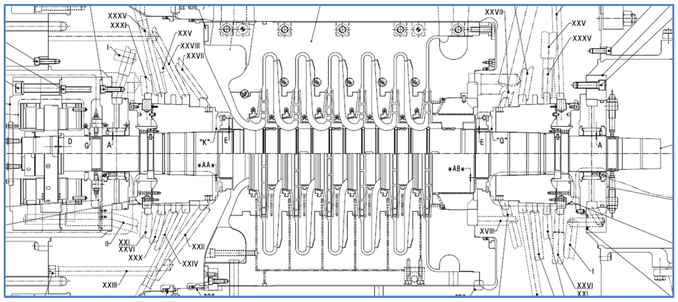

Figure 1: BCL306/C cross section

The identification is performed on three different operating points of the compressor representative of the most significant conditions.

To underline the reliability of OMA methodology some numerical cases are also presented. Aim of those numerical cases is to provide some assessment of OMA's accuracy and to draw some valuable general considerations for the OMA analysis.

OPERATIONAL MODAL ANALYSIS OVERVIEW

OMA is a technique for modal parameter estimation that does not require the knowledge of the input loading force. In OMA, the excitation forces are unknown. If, for those input forces, a Gaussian white noise distribution is assumed, then the output spectrum is fully representative of the structure and modal parameters can be identified. Unfortunately, this assumption is not correct for a real operating centrifugal compressor, because there are always excitations at some frequencies containing more energy than other. The observed response is the combination of the natural vibration of the rotor and the excitation mechanism as reported in Figure 2. Poles related to both structure and excitation are identified all together but thanks to the Singular Value Diagrams and the statistical properties of the identified mode, it is possible to recognize and distinguish an excitation from a mode of the rotor system.

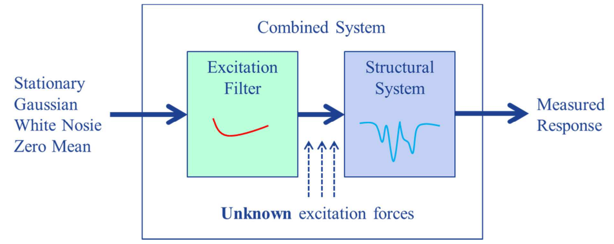

Figure 2: Schematic representation of OMA model

During the performance test of a centrifugal compressor, radial proximity probes raw signals were recorded at steady state operating conditions. These data over a certain time period are needed to perform the operational modal analysis of the compressor.

By means of the Stochastic Subspace Identification (SSI) algorithm, Brinker and Andersen (2006), it is possible to estimate the modal parameters of the system.

SSI is a "black-box" multiple degree of freedom algorithm performing the fitting of the response of a generic structure with a linear equations system. Basically it is able to identify the state space matrix of the system in its linearized formulation and therefore the poles. For each pole the associated eigenvector is also plotted using the axial location of radial proximity probes.

The use of OMA for logarithmic decrement evaluation on centrifugal compressors was validated at the authors' Company through a direct comparison with a traditional stability test employing a magnetic exciter, confirming the reliability of its results (Guglielmo et al., 2014). The broad-band excitation source applied to the rotor is represented by the aerodynamic force of the inlet gas flowing into the compressor suction. This force in a high pressure centrifugal compressor tested at full load conditions is large enough for the purpose of the analysis.

Differences and similarities between EMA and OMA techniques

In traditional experimental modal analysis (EMA), where excitation forces are artificially applied and measured, the modal parameters of structures are extracted from frequency response function (FRF) for frequency domain techniques or impulse response function (IRF) for time domain techniques.

Assuming a white noise excitation the auto and cross correlation functions of the response contain the same information of impulse response functions, making similar the problem of modal parameters extraction in EMA and OMA methods. Overschee et al. (1996) present examples with both deterministic and stochastic subspace identification problems and algorithms.

Thus the main difference between EMA and OMA is not in the modal parameters extraction but in the excitation source. It is important to highlight that the same sort of problems related with the process of modal parameters estimation is present in both, OMA and EMA.

The main advantage of OMA is the absence of external excitation device, which makes stability tests a simpler task and also makes possible to assess compressors stability during its field operation. One drawback is that OMA relies in ambient excitation, which could result in two problems: a low amplitude response and an excitation different from the ideal white noise (zero mean, stationary Gaussian distribution). The other disadvantage is that as the input force is not known the amplitude of modal response cannot be scaled for the operational case.

EMA helps to ensure that the mode of interest will be excited, thanks to a controlled known excitation. Unfortunately is needed the addition of a temporarily external excitation device, which changes the system dynamics (although OEMs minimize this as much as possible), moreover additional costs and additional lead-time on the project have to be taken into account.

HARDWARE

The compressor showed in Figure 1 was modified to allow the installation of the magnetic exciter; in particular the axial bearing housing has been modified. The magnetic exciter was temporarily mounted on the non-drive. A shaft extension with the lamination stack assembled on it has been connected by a tie rod.

The maximum force that the magnetic exciter can apply is 1200 N, the force can be just in one direction or, by proper tuning of the phase lag, is possible to obtain an oscillating force or a rotating constant force. The feedback signals for the closed loop control are given by a sensor ring adjacent to the control axes.

As usual, the machine is equipped with proximity probes at bearings location to monitor radial vibrations. The additional measurement plane at sensor ring was also available for model identification.

TEST RESULTS

In order to validate the OMA method, a comparative analysis with traditional stability test was performed at different pressurization levels of gas loop with the compressor running at MCS.

Figure 3 shows the position in the compressor map of the operating conditions explored during the stability test. Excluding the first test all measured points were characterized with the same pressure ratio and different pressure level as reported in Figure 4.

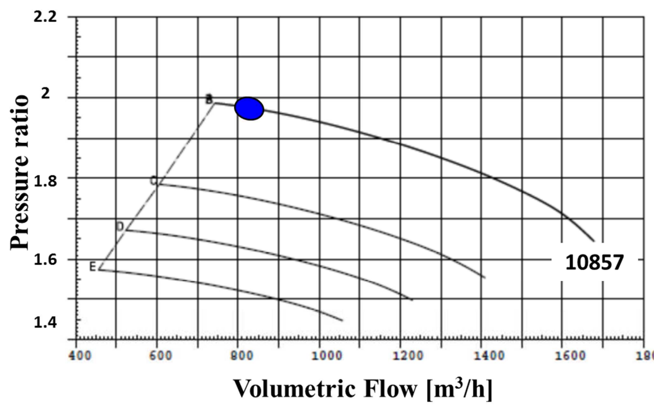

Figure 3: Operating condition for stability test

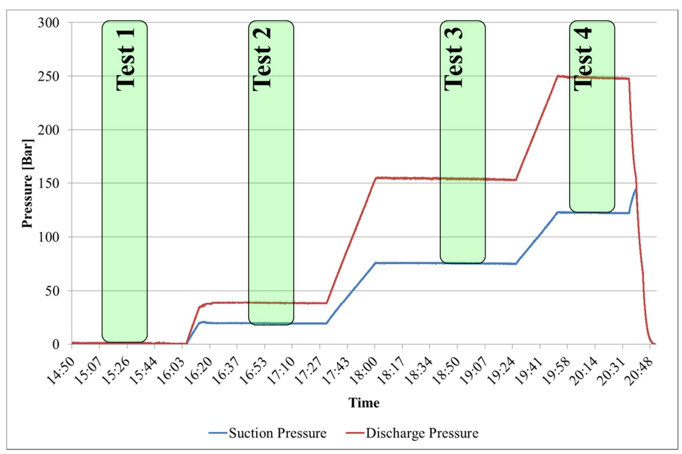

Figure 4: Pressure levels trend

The first test was performed at very low differential pressure, around 1 bar, in order to minimize the damping effect of the honeycomb seal and to reproduce the behavior of rotor plus bearing system. The inlet pressure was set at the minimum value allowable by dry gas seal system using an external source of clean gas to provide the differential pressure required for the sealing.

Compressor operating point was close to the choke and the thermodynamic steady state condition has been reached before starting to excite the rotor. A directional excitation, respectively horizontal and vertical, was applied to identify the forward mode for each test point. The continuous sine sweep signal with a very slow sweep rate has been used to excite the rotor and an order tracking analysis is done to extract the frequency response functions for the four probes. The force is evaluated by knowing the coils current.

A MDOF fitting algorithm in frequency domain like PEM (L. Ljung) was used to post process the frequency response functions. A selection of response channels, driven by the value of SNR, was performed before starting the identification process. Typically the fitting was performed using 4 channels simultaneously.

For the same operating condition, an OMA analysis was performed by using all the steady state vibration signals. In this case all six available channels were used for the fitting.

Figure 5 shows the experimental data and the fitted FRF at NDE bearing obtained with AMB. In Figure 6 is reported the SVD plot of the output matrix for the OMA analysis. Both methods work very well with forward mode. In case of OMA, where no artificial excitation is used, the detection of backward mode is weaker and usually its evidence in the response spectrum depends on the level of excitation and damping of the system.. In this case, at low pressure, the backward mode was identified pretty well and it is highlighted by the simultaneously modulation of two Singular Values. Further details on the useful information that can be extracted from the SVD plot are reported in Guglielmo et al., (2014).

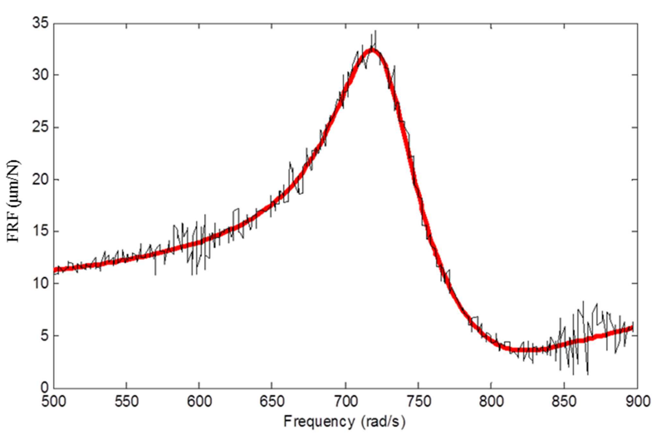

Figure 5: PEM FRF at NDE for Case 1

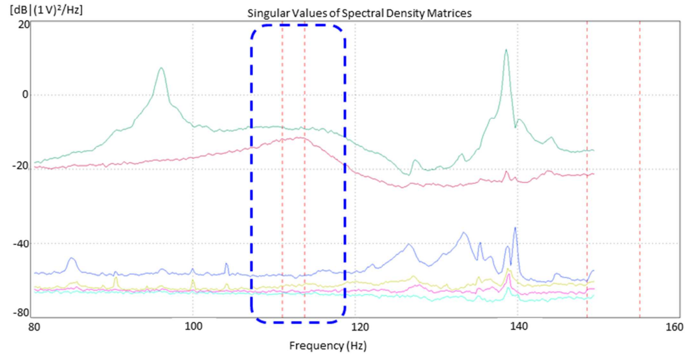

Figure 6: OMA SVD plot for Case 1

The results of this first comparison are reported in Table 2 with a good agreement between two methods. For Case 1, which is a light damping case, there is a discrepancy below the 2%.

		AMB		OMA		Difference	
Case	Suction Pressure [bar]	Frequency [Cpm]	Modal Damping [Log Dec]	Frequency [Cpm]	Modal Damping [Log Dec]	Frequency	Modal Damping
1	1	6949	0.246	6984	0.25	-0.5%	-1.6%
2	19	7260	0.513	7470	0.525	-2.9%	-2.3%
3	75	8296	1.52	8568	1.225	-3.3%	19.4%
4	120	8974	2.4	n.a.	n.a.	n.a.	n.a.

Table 2: Identification results summary

In Case 2 the suction pressure of the compressor was increased at 19 bar and the operating point was moved close to anti surge with a pressure ratio around 2.

Also, in this case, the position of forward mode is well identified by both methods with an overall reduction of amplitude vibrations due to the increased damping level (Figure 7 and Figure 8). The error between the two methods is still acceptable with a maximum difference below 3%.

The Case 3 of the comparison between PEM and OMA was performed with an inlet pressure of 75 bar with the same pressure ratio. Due to the presence of honeycomb seal on the compressor balance drum the damping of the first mode increases with the pressurization level and, for both methods, the response is quite flat (Figure 9 and Figure 10) with no evidence of clear peak in the response.

In this case the results of identification process are sensitive to the fitting parameters and also the repeatability of each test shows a discrepancy on the logarithmic decrement estimation between the two methods around 20%.

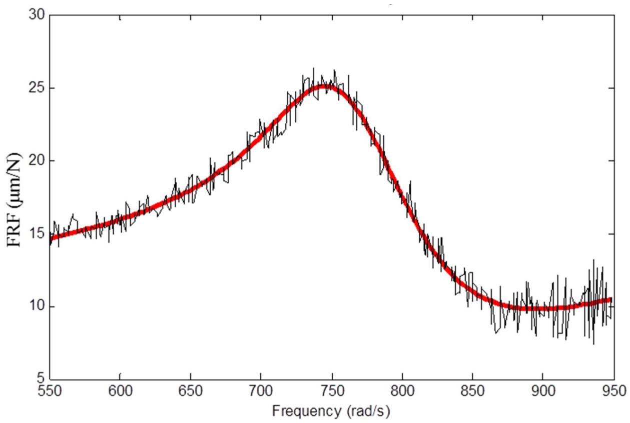

Figure 7: Case 2 - PEM FRF at NDE for Case 2

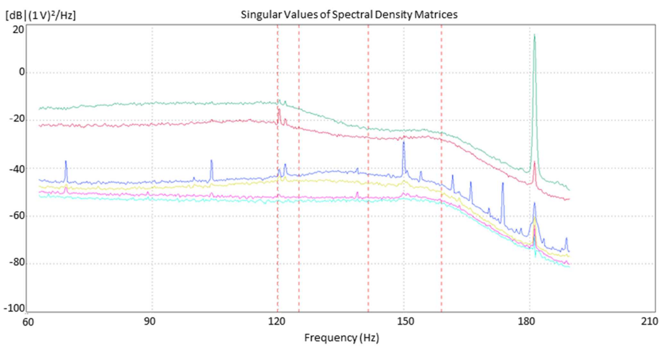

Figure 8: OMA SVD for Case2

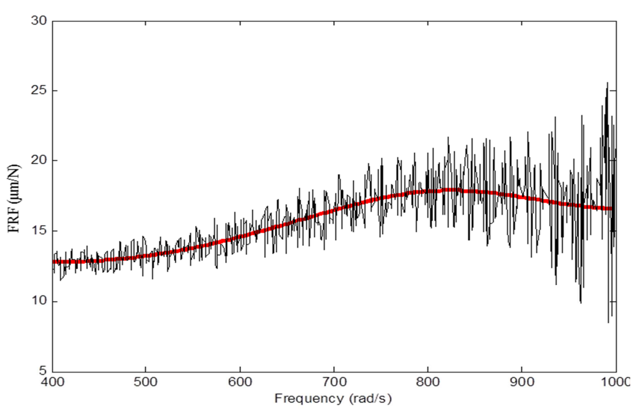

Figure 9: PEM FRF for Case 3

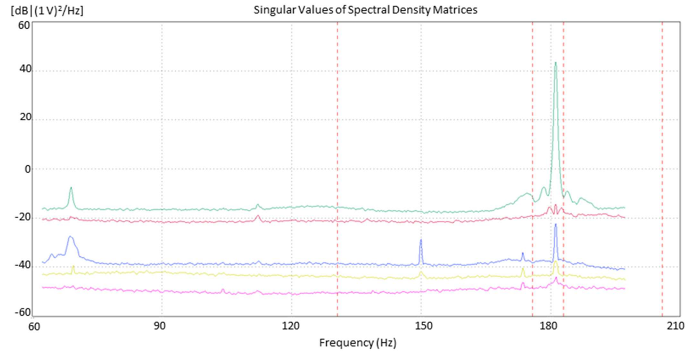

Figure 10: OMA SVD for Case 3

Finally, in Case 4, the suction pressure was increased up to 120 bar and the stability measurement was repeated. In this case, no strong evidence of excited mode is present in the FRF of all probes. The response spectrum, reported in Figure 11, is flat and identification of damping is very challenging for both methods.

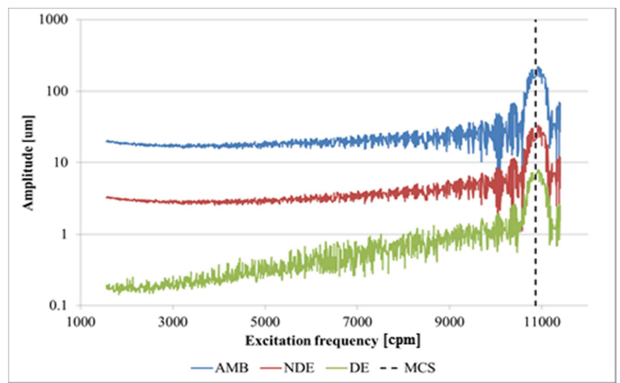

Figure 11: FRF for Case 4

PEM algorithm still identifies a mode close to 8970 rpm with a log dec of 2.4 but the results are not reliable. At each step of pressurization five runs of excitation sweep were collected, in order to allow evaluation of repeatability. No results can be achieved with OMA, which shows no vibration activity in the frequency range of interest.

According to the results reported in Table 2 and

Figure 12 both methods provide reliable results in when log dec value is lower than 1, with a maximum deviation between them lower than 3% and with a fairly good repeatability.

With log dec higher than 1 both methods suffer in the evaluation of damping with significant deviation between EMA and OMA but also with the absence of strong repeatability in the identification results.

Despite so these differences are not so important from a stability perspective due to the high absolute value of log dec.

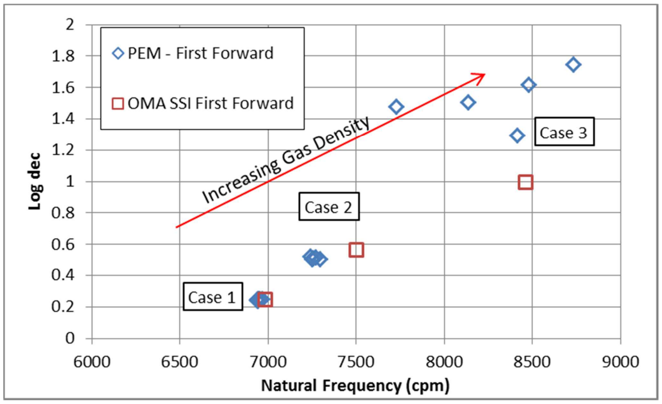

Figure 12: Comparison between the results of EMA PEM and OMA SSI for compressor stability test

NUMERICAL TEST CASE FOR OMA APPLICATIONS

Despite the OMA is a well-known technique used in civil and aerospace structures, its application for rotordynamic purpose in Oil&Gas world is pretty new and without strong references. There are several factors that make the use of OMA not always intuitive in rotordynamics, here the importance of an expert test engineer to ensure the reliability of the results.

One factor to consider is that modes of a centrifugal compressor system appear generally in a couple of modes, often with very close frequencies (if there isn't a strong anisotropy) and with different level of damping depending on operating conditions or by the presence of damper devices.

An additional factor that can make the use of the OMA more challenging is the level and type of excitation forces acting on the rotors. Especially in centrifugal compressors, where a standard aerodynamic design together with robust structures is employed, the level of excitation sources is reduced and it clearly affects the level of response. For the same reasons the type of excitation (very different from a wind storm acting in a building or a bridge) makes in some case the identification very difficult (pseudo stationary with the presence of several harmonics).

Moreover, the level of observability of the system is another key point. In the compressor standard configuration, the vibration sensors are usually placed just in two planes close to the bearing where the amplitude of the first mode can be very low, especially if very stiff bearings are used.

For all of the above mentioned reasons, the interpretation and accuracy of OMA results is not a straightforward process and often the judgment of the test engineer is necessary for the validation of results. The signal post-processing phase plays a fundamental role in the modal identification process and the detection of main modal indicators is a robust support for the validation of curve fitting.

In the present paper, the authors present an overview of practical cases that can provide useful guidelines for the interpretation of OMA results.

A finite element model of a compressor was used to run a transient time response for a given excitation and the results were used as input for OMA identification.

The model features two orthotropic bearings and one seal placed in the mid span of the rotor.

The rotor model is excited with a white noise filtered with a low pass filter set up well above the rotor modes under investigation. In the excitation used in the light damped case it was also added the unbalance force (harmonic).

The properties of bearings and seals are artificially changed in order to have modes with different level of damping and/or frequency.

An additional uncorrelated noise with broadband is added to the vibration signals to simulate the measurement noise of a real case.

The time history (60 seconds length) is then analyzed with OMA in the same way of experimental data and the relation between the changes of modal indicators with different modal properties of rotor is investigated.

The compressor rotor model is represented in Figure 13.

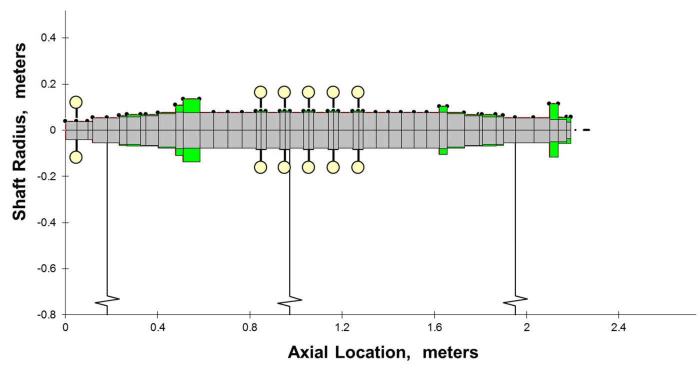

Figure 13 Compressor model

Test case 1- Rotor with marginal level of damping

During the modeling phase, cross-coupling coefficients of the seal were modified in order to create a forward mode with a log dec close to zero leaving the damping level of backward mode set to a higher value. Both OMA and PEM identifications were performed for this same model. A 200 N force was applied to the last right node in the vertical direction simulating a stepped sine sweep. EMA is able to identify properly both FW and BW frequency and logarithmic decrement independently from the damping level, it has to be highlighted that noise was not added to the simulated responses. The state space model size was varied in order to recognize the real modes of the system: these modes shall remain reasonably the same in terms of log dec and frequency.

A segment of the response waveform is reported in Figure 14 for the two probes. The results of the simulation are reported in Table 3 while the results of the identification are reported in Table 4.

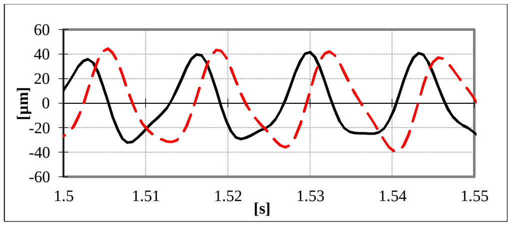

Figure 14 Response waveforms for the light damping case at bearing location in the two orthogonal directions

Rotor	Frequency [CPM]	Log Dec	Modal Damping [%]
1st BW mode	4302	0.34	5.4
1st FW mode	4392	0.0001	0.002

Excitation	Frequency [Hz]	Type
1st BW mode	1666	White noise
1st FW mode	600	Low-pass
Harmonic	152	1X
Speed	152	9120 RPM
Noise/Harmonic amplitude	10%	

Simulation	
Length [s]	60
Measurement noise	2%

Table 3 Modeling - Light damping case

	Numerical model		OMA		EMA PEM	
	Frequency [CPM]	Log Dec	Frequency [CPM]	Log Dec	Frequency [CPM]	Log Dec
1st BW mode	4302	0.340	4298	0.330	4301	0.341
1st FW mode	4386	0.00013	4362	0.00025	4389	0.00018

Table 4 OMA and EMA Results - Light damping case

OMA SSI algorithm works well with the backward mode providing an error in the estimation below 2%. For the forward mode where the damping is very low, the percentage difference in the damping estimation is high but negligible from an engineering point of view (the mode is anyway very close to instability).

Theoretically, the SSI algorithm isn't able to discriminate the difference between harmonic components and rotor modes with logarithm decrement close to zero. Everything included in the observed data is fitted with the state space model, so a marginal stable rotor is seen like a forcing phenomenon exciting the structure continuously at the same frequency. There are two basic features helping in the harmonic recognition (which is different from a no damped modes) and these can be used in the post processing phase of vibration signals.

Rotordynamic modes appear in couples of modes that normally are both excited by the noise with different energy levels. Therefore, the SVD plot will show the modulation of two Singular Values with "bells" that reflect the level of associated damping. In case of a mode with a very small damping, the peak of first SVD will look like one of a harmonic but beside it the modulation of second Singular Value (red line) allows the recognition of mode response. For the same reasons, harmonics will be usually recognized in the SVD plot with a peak in all SVD (output matrix *n* times singular) and without any smooth modulation.

Figure 15 clearly depicts the difference between the un-damped mode at 73 Hz and the harmonic (in this case the rotor speed is at 150 Hz).

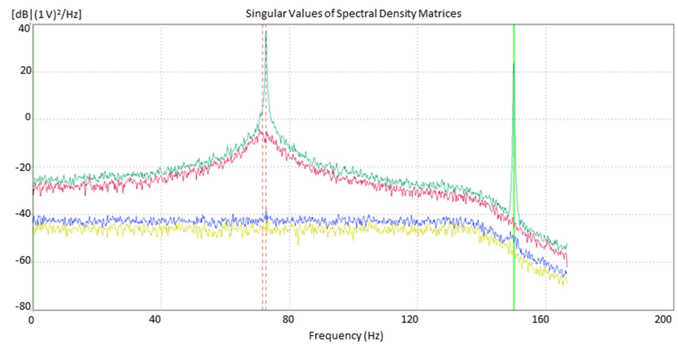

Figure 15 SVD plot - Light damping case

Another parameter that discriminates the response of an un-damped mode from forced response peak is the value of Kurtosis.

Both types of responses appear in the spectrum with a tone at a well-defined frequency. A method to distinguish between a harmonic and a mode is to look at the difference that can be found in the coherence length (the time for which a signal maintain the same amplitude and phase relationship with itself), if the peak is related to a mode the coherence length is short.

On the other hand, the forced response signal, i.e. the speed, will maintain the same amplitude and phase unless the rotor condition changes. This difference in the coherence length between two types of response can be quantified using the kurtosis estimator, here denoted with γ.

$$k = (x|\mu,\sigma) = \frac{E[(x-\mu)^4]}{\sigma^4} \qquad (1)$$

The response spectrum is band pass filtered and for each frequency window the time response is reconstructed and kurtosis is evaluated.

The kurtosis is defined as the fourth central moment of the probability density function normalized with respect to the standard deviation σ.

For Gaussian distributed measurements, $\gamma = 3$, whereas for perfect sinusoidal distributed measurements $\gamma = 1.5$.

In the case of undamped rotor, the kurtosis exhibits two local minimums at mode frequency and at speed frequency while the overall value is above 3.

As can be seen in Figure 16, the kurtosis reaches a value of 1.5 at the frequency of excitation, representing a perfect sinusoidal signal stable over the measurement time. At the mode frequency the reached value is 1.8 mainly due to the amplitude and phase variation.

According to the authors experience, consolidated with the analysis of several tests carried out on compressor test vehicles, the value of kurtosis for mode with log dec lower than 0.05 is still above 3.

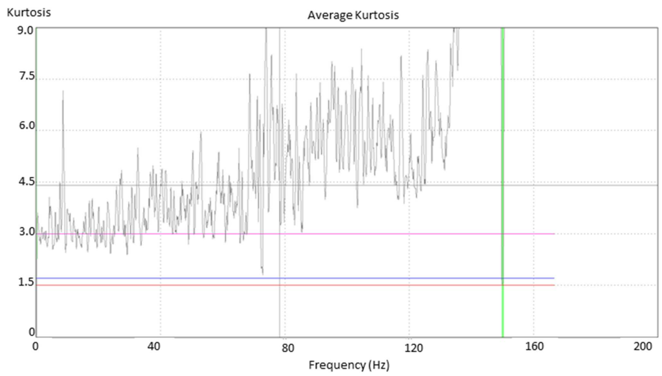

Figure 16 Kurtosis plot for light damping case

Test case 2- Rotor with increasing damping level

The opposite situation of the previous case is a compressor with a high level of damping. Generally, for high pressure application, compressors are equipped with damper devices that increase the level of damping (a well-designed Honeycomb seal is an example).

In this case the spectrum of the rotor response in both cases of traditional stability test and OMA is usually very flat and both methods could suffer in the accuracy of the identification of modal properties. The results of the identification process can be no reliable and in many cases are dependent on the curve fitting parameters.

For test case 2 the seal dynamic force coefficients are modified to get higher and higher logarithmic decrements of the first forward mode.

Rotor	Case	Frequency [CPM]	Log Dec	Modal Damping [%]
1st BW mode	a	4302	0.189	3.0
	b	4272	1.123	17.6
	c	4212	1.553	24.0
1st FW mode	a	4386	0.151	2.4
	b	4356	0.492	7.8
	c	4296	1.051	16.5

Excitation	Frequency [Hz]	Type
1st BW mode	1666	White noise
1st FW mode	600	Low-pass
Harmonic	NA	NA
Speed	210.4	12625 RPM
Noise/Harmonic amplitude	NA	

Simulation	
Length [s]	60
Measurement noise	2%

Table 5 Modeling Results for an increasing damping level

As can be seen in Table 5, for these cases the noise excitation was applied without any synchronous component superimposed. The response waveform at one bearing for the *Case a* is reported in Figure 17.

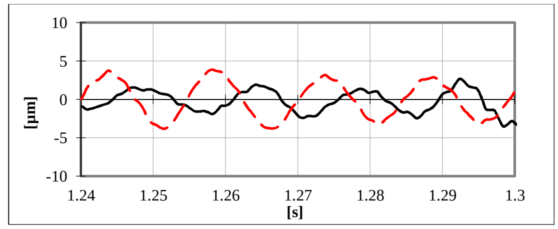

Figure 17 Response waveforms for Case a at bearing location in the two orthogonal directions

The response spectrum becomes quite flat with respect to the light damping case, with consequently reduction of the gap between the first and the last singular value. Figure 19 to Figure 20 show the modification of SVD plot with an increasing level of damping.

In this situation, the energy level of structure response for both modes is close to the level of uncorrelated noise and the results of the identification process are affected by the fitting parameters set up.

The results of modal identification are reported in Table 6. The OMA frequency prediction accuracy is quite good with an error below 5%.

Also the modal damping is estimated with a good accuracy up to a level of log dec ~1.0 for both forward and backward mode. Above this value, the error provided by OMA is about 20% but at the same time the importance of absolute value of logarithmic decrement decreases.

	Case	Numerical model		OMA		EMA PEM	
		Frequency [CPM]	Log Dec	Frequency [CPM]	Log Dec	Frequency [CPM]	Log Dec
1st BW mode	a	4302	0.189	4277	0.180	4300	0.188
	b	4272	1.123	4329	1.040	4269	1.122
	c	4212	1.553	4142	1.820	4213	1.551
1st FW mode	a	4386	0.151	4369	0.150	4387	0.149
	b	4356	0.492	4356	0.450	4353	0.494
	c	4296	1.051	4412	0.950	4293	1.049

Table 6 OMA and EMA results for an increasing damping level

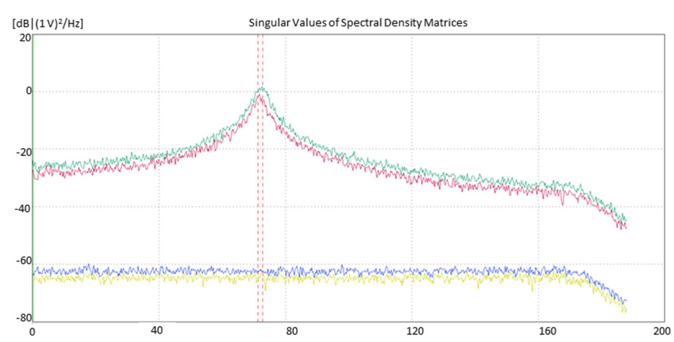

Figure 18 SVD plot - Case a

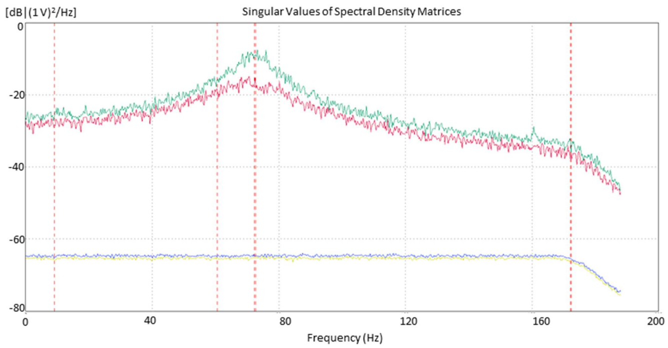

Figure 19 SVD plot - Case b

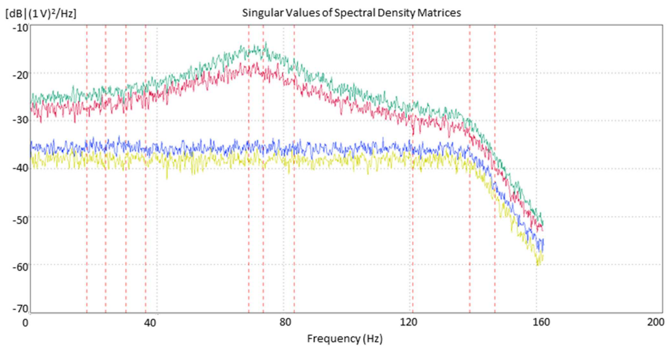

Figure 20 SVD plot - Case c

A useful indication about the frequency of modes is also provided by the coherence plot. The coherence function between two random signals, here defined in (2), gives an indication about the relation between them evaluated over the measurement time, where $S_{xx}(\omega)$ and $S_{xy}(\omega)$ are the auto power spectrum and cross power spectrum of two random single x(t) y(t).

$$\gamma_{x,y}(\omega) = \frac{|S_{xy}(\omega)|^2}{S_{xx}(\omega)S_{yy}(\omega)} \tag{2}$$

In traditional EMA, the coherence is calculated between the excitation force given to the structure and the generated response. A high value of coherence means that the measured response is generated by the injected force, therefore the quality of identified frequency response function is guaranteed.

In OMA, the coherence is calculated between the different measurement points. In this case, a high value of coherence means a stable ratio between different probes and normally it happens in the presence of excited mode or harmonic. In case of a harmonic, the entire structure is forced to vibrate with a specific shape. In case of a mode, if the level of uncorrelated noise doesn't overcome the structure response, the amplitude and phase relation between two points is constant. This amplitude ratio reflects the mode shapes amplitude.

Therefore, the frequency of modes can be highlighted by an increase of coherence function between the vibration signals at the two sides of rotor as showed in Figure 21 and Figure 22.

In this frequency region, a stable relation between the two sides of the rotor means that the response is driven by the presence of couple modes randomly excited by noise but with constant amplitude ratio given by the associated mode shapes. Outside of this region the rotor response is driven by the excitation force and the coherence function drops down.

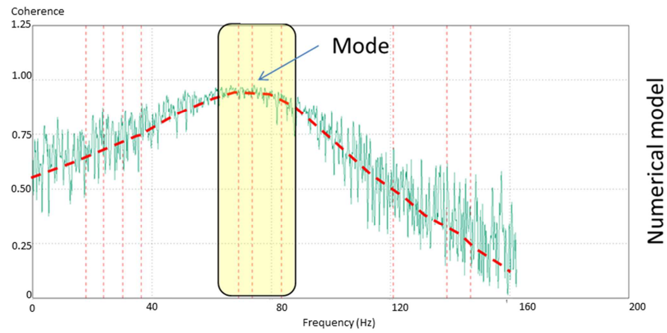

Figure 21 Coherence function for numerical model

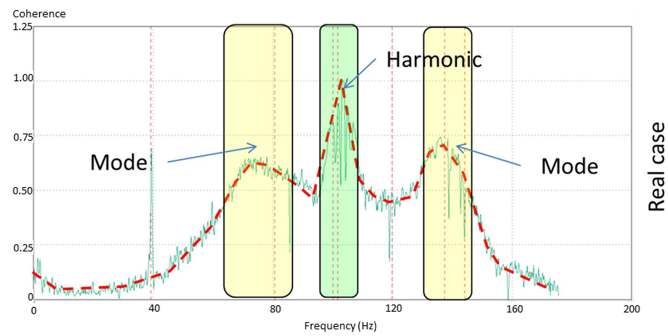

Figure 22 Coherence function with harmonic in real compressor case

Test case 3-Rotor with baseplate dynamic interaction

Very often today turbomachines are required to be fast, efficient and compact. This design philosophy also applies to baseplate especially for off shore application where the weight reduction is necessary leading in some case to have a more flexible structure.

From an analytical perspective, many modelling techniques have been developed in the past years in order to take into account structure effects that cannot be considered rigid.

From an experimental point of view, the use of OMA provides a quick and easy instrument for the identification of baseplate dynamic behavior just using some accelerometers placed on the structure. If the only available probes are the rotor probes, the recognition of the baseplate dynamics from the rotor response becomes more challenging.

From OMA point of view, in principle, there isn't any difference between pure rotor modes and coupled modes with interaction between rotor and baseplate.

In many cases coupled modes can be recognized among the set of identified modes by looking at their characteristics.

If the motion is driven by a decrease of dynamic stiffness of baseplate, the main quantity of kinetic energy associated to the modes is located at the bearing supports with low relative amplitude at bearing oil film. Therefore, the main damping source of the coupled mode is the structural damping of baseplate leading to a modal damping typically lower than ~2%.

Another feature that helps in the recognition of rotor-baseplate interaction is the shape of couple modes.

Since the baseplate is a non-symmetric structure, the identified system modes (rotor + baseplate) are often uncoupled and spaced in frequency due to the different stiffness level in the vertical and horizontal planes. This is reflected in the shape of coupled modes that generally appear as non-rotating modes and in many cases with motion localized at one bearing.

Finally, in case of dynamic interaction with baseplate, the shape of the rotor response is qualitatively different from the one of rotor only. The frequency band with dynamic interaction between the two structures appears in the spectrum as discontinuity of a main Singular Value represented by the response of rotor alone.

This phenomenon is typical of structures with different dynamic behavior and coupled together on few connection points.

Modification of rotor response function can be qualitatively explained using the basic formula of dynamic structure interaction expressed in terms of dynamic stiffness.

In (3), the suffix C accounts for the coupled system and R indicates the rotor alone; i and j are two points of the rotor and c is the connection point.

$$K_{i,j}^{C}(\omega) = K_{i,j}^{R}(\omega) + \frac{K_{i,c}^{R}(\omega)\cdot K_{c,j}^{R}(\omega)}{K_{cc}^{R}(\omega)+K_{cc}^{Base}(\omega)} \qquad (3)$$

Out of the resonance frequency of the baseplate, dynamic stiffness $K_{cc}^{Base}(\omega)$ is high and the second term of equation 3 is close to zero. Therefore, the dynamic behavior of coupled structure $K_{i,j}^{C}(\omega)$ is equal to $K_{i,j}^{R}(\omega)$.

When the excitation frequency approaches baseplate natural frequency, second term of equation (3) becomes significant and the two systems are strongly coupled.

Also, in this case, the finite element model was used to reproduce a time history that can be processed with OMA. The rotor model was modified by adding a virtual beam of M mass that supports the bearings and connected to the ground with a spring damper element. The values of M, C, and K were chosen to set the resonance frequency of baseplate at the specified value for the simulation. The results of simulation are reported in Table 7.

Rotor	Frequency [CPM]	Log Dec	Modal Damping [%]
1st BW mode	4339	0.175	2.79
1st FW mode	4458	0.161	2.56
Baseplate mode	2050	0.139	2.21

Excitation	Frequency [Hz]	Type
1st BW mode	1666	White noise
1st FW mode	600	Low-pass
Harmonic	NA	
Speed	210.4	12625 RPM
Noise/Harmonic amplitude	NA	

Simulation	
Length [s]	60
Measurement noise	2%

Table 7 Modeling Results for the case with supports effect included

The results of OMA identification are reported in Table 8. The algorithm provides an estimation of frequency and damping of modes also in this case with an error lower than 2%.

	Numerical model		OMA		Error	
	Frequency [CPM]	Log Dec	Frequency [CPM]	Log Dec	Frequency	Log Dec
1st BW mode	4339	0.175	4318	0.170	-0.5%	-0.8%
1st FW mode	4458	0.161	4442	0.160	-0.3%	-0.7%
Baseplate	2050	0.139	2047	0.137	-0.2%	-1.7%

Table 8 OMA results for the case with supports effect included

In this case, the SVD plot reported in Figure 23 provides a graphical representation of the area interested by the interaction between rotor and baseplate.

At the resonance frequency of the baseplate, the vibration is dominated by a coupled behavior. Moving away from resonance, the influence of baseplate drops down from first to last Singular Value while the influence of rotor behavior on vibrations rises up, dominating the first two Singular Values for the whole frequency range as can be seen in Figure 24.

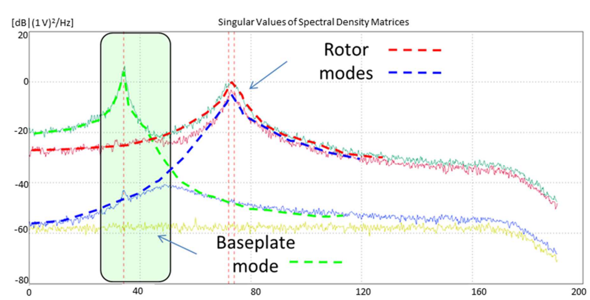

Figure 23 Mode shapes for the case with supports effect included

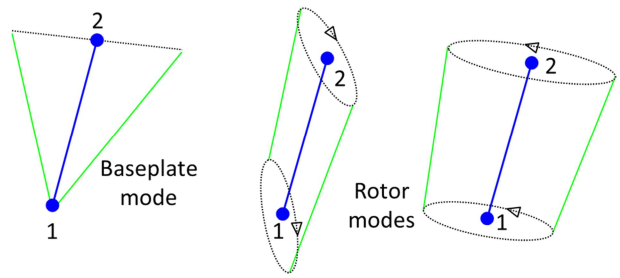

Figure 24 SVD plots for the case with supports effect included

CONCLUSIONS

OMA is a really powerful methodology if properly used; rotordynamics and modal testing knowledge are strongly required to get a reliable post-processing.

The comparison of experimental identified modal parameters of a particular high pressure compressor tested at full load highlights the equivalence of conventional stability test carried out through the magnetic exciter and the Operational Modal Analysis. A very good agreement between the two testing techniques in the results was obtained together with the advantage of OMA with respect to traditional stability test in avoiding rotor modification required for the installation of magnetic exciter.

The paper showed also the reliability of OMA methodology for the estimation of the stability level of the first lateral mode of centrifugal compressors. By the creation of finite element model of a centrifugal compressor, it was possible to perform a simulation of a stability test carried out by OMA. The results of the identification obtained are very close to the theoretical values given by eigenvalues analysis of the model especially for lightly damped modes. For the overly damped modes, OMA suffers with repeatability and reliability, although those modes do not represent a concern from the engineering perspective. Moreover, some tips and hints were delivered to the reader to better understand the difference of a quasi-unstable mode from a harmonic excitation or to recognize a structural resonance from rotor modes

Despite the successful application of OMA techniques on these cases, some concerns are worthy to be enumerated in the use of this technique.

The main assumption of OMA is the presence of a noise excitation for the detection of rotor modes, but in case of low pressure applications where the gas flow does not provide excitation strong enough, this assumption could not be verified.

At the same time, the non-controlled excitation in OMA analysis can results in a lot of modes in the response spectrum, making challenging and complex the identification process of rotor modes and, more in general, providing a set of results valid just for the tested conditions.

Despite so, the authors will continue to invest in the application of this methodology considering OMA a reliable alternative to AMB for the stability measurements.

NOMENCLATURE

C	= Damping
K	= Stiffness
$K_{i,j}$	= Dynamic Stiffness
M	= Mass
γ	= Coherence function
κ	= Kurtosis function
ω	= Pulsation
AMB	= Active Magnetic Bearing
BW	= Backward
EMA	= Experimental Modal Analysis
FWD	= Forward
FRF	= Frequency Response Function
Log dec	= Logarithmic Decrement
MDOF	= Multi Degrees Of Freedom
ME	= Magnetic Exciter
OEM	= Original Equipment Manufacturer
OMA	= Operational Modal Analysis
PEM	= Prediction Error Method
SSI	= Stochastic Subspace Identification
SVD	= Singular Values Decomposition
S_{xy}	= Cross power spectrum function
S_{xx}	= Auto power spectrum function

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